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Comparison of Predicted and Measured Elastohydrodynamic Film Thickness in a 20-Millimeter-Bore Ball Bearing

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SUMMARY

Elastohydrodynamic film thicknesses were measured for a 20-mm bore ball bearing using the capacitance technique. The bearing was thrust loaded to 90, 445, and 778 N (20, 100, and 175 lb). The corresponding maximum contact stress on the inner race was 1.28, 2.09, and 2.45 GPa (185 000, 303 000, and 356 000 psi). Test speeds ranged from 400 to 15 000 rpm. Measurements were taken with four different lubricants: (1) synthetic paraffinic, (2) synthetic paraffinic with additives, (3) synthetic type II aircraft engine oil meeting MIL-L-23699A specifications, and (4) synthetic cycloaliphatic hydrocarbon traction fluid. The test bearing was mist lubricated. Test temperatures were 27°, 65°, and 121° C (80°, 150°, and 250° F).

The measured results for the various test parameters were compared to theoretical predictions from computer programs. Also the data were plotted on dimensionless coordinates and compared to several classical isothermal theories. Agreement with the computer program results was only fair. Agreement was best at low speeds and higher temperature.

Plotting the results on dimensionless coordinates unified the data. There was good agreement at low dimensionless speed, but the film was much thinner than theory predicts at higher speeds. It is shown that inlet shear heating partially accounts for the film thickness reduction, whereas starvation is shown to be the primary factor.

INTRODUCTION

The elastohydrodynamic (EHD) lubrication of bearings has been investigated theoretically and experimentally for more than 20 years (refs. 1 to 16). Two significant effects occur in EHD lubrication, namely, the influence of high pressure on the viscosity of the liquid lubricant, and substantial local deformation of the elastic contacts. The net result is the formation of a thin lubricant film that is beneficial in preventing seizure and rapid wear of contacting parts.

It is important that engineering data on EHD film formation be accurate and comprehensive for proper evaluation and application of theory. Experimental measurement of film thickness is very difficult because the films are thin. Various methods have been developed and used: X-ray, optical, and electrical capacitance and conductance (ref. 17). X-ray measurements are only suited to measure film thickness between contacting disk rollers. Optical methods have been generally applied to the case

of a steel ball on a transparent disk. However, a recent application was made to a radially loaded roller bearing with a quartz window in the outer race (ref. 18). In reference 19 it was concluded that film thickness measurement using the conductance method is not practical because there is a transition from no contact to full contact over a very small range of film thickness. Of the various methods available, only the capacitance method is suitable to measure film thickness in an operating rolling element bearing because of the heavy loads and geometric complexity involved. To date, no film thickness data has been available for a rolling element bearing operating under conditions of heavy load and high speed.

The objective of the research described in this report was to measure lubricant film thickness in an operating ball bearing for various conditions of temperature, load, and speed, for four lubricants. A second objective was to compare the results to theoretical predictions and determine the general applicability of existing theories to a practical situation.

In order to accomplish the above objectives elastohydrodynamic film thicknesses were measured for a 20-millimeter-bore ball bearing using the capacitance technique. The bearing was mist lubricated and run at speeds ranging from 400 to 15 000 rpm. The bearing was thrust loaded to 90, 445, and 778 newtons (20, 100, and 175 lb), giving maximum contact stresses on the inner race of 1.28, 2.09, and 2.45 GPa (185 000, 303 000, and 356 000 psi). Four different lubricants were used at three different test temperatures: 27°, 65°, and 121° C (80°, 150°, and 250° F).

APPARATUS, SPECIMENS, AND PROCEDURE

EHD Film Thickness Test Rig

The experiments were conducted using the test rig shown in figure 1. The test rig consists of a rotating driveshaft supported by two bearings running against a test bearing which is located in a vibration isolated housing. The shaft is driven by an air turbine. Shaft speed is sensed with a magnetic pickup located in close proximity to six equally spaced slots milled into the shaft. The speed is held constant during a test with an automatic controller. The load is applied pneumatically. The linear ball bushing allows the thrust load to be transmitted from the pneumatic loader through the shaft to the test bearing. The thrust load is determined from the air pressure acting on the loading cylinder area. The temperature of the test bearing is controlled by cartridge heaters which are located in the test bearing support housing. The temperatures were held constant within approximately 3° C (5° F) during a test using an automatic controller on the heaters. Temperatures were sensed by thermocouples placed on the bearing

outer race and in the inlet lubrication stream. The bearings were mist lubricated with separate systems for the test and support bearings. A separate heater control on the inlet mist system permitted the test bearing lubricant temperature to be held equal to the bearing outer race temperature.

Test Bearing

The test bearings for this experiment were 20-millimeter-bore angular contact bearings having three 7.15-millimeter (9/32-in.-) diameter balls. The bearing specifications are given in table I. The bearing had a polyimide cage and the outer race was electrically insulated from the housing. The spring-loaded thermocouple at the outer-race and the slip-rings on the drive shaft provided an electrical contact for capacitance and conductance measurements across the test bearing.

Test Lubricants

Four different lubricants (see table II) were used in the film thickness experiments. The lubricant properties are given in table II. Lubricants A and B were experimental synthetic paraffinics with similar properties, but B had antiwear additives whereas A did not. Lubricants A and B were used because they had been used in some experiments conducted on the roller disk machine at the Battelle Memorial Institute (ref. 12). Lubricant C was an improved type II synthetic oil meeting the MIL-L-23699 specification. It is an aircraft turbine engine oil and is also used in engine gear boxes and helicopter transmissions. Lubricant D was an experimental traction fluid (synthetic cycloaliphatic hydrocarbon), used in traction drive transmissions. Lubricants C and D have similar temperature-viscosity properties; lubricant D has a larger pressure-viscosity exponent.

Measurement Scheme

The lubricant film thickness was measured using the electrical capacitance method as described in references 3 and 6. The electrical capacitance between the inner and outer races of the test bearing is a function of the thickness of the lubricant films and the sizes of the Hertzian areas at the inner and outer race ball contacts (table IV). The capacitance across the test bearing was measured with a capacitance bridge and the film thickness was read from the calibration plot shown in figure 2. In calculating the capacitance values plotted in figure 2, it was assumed that inner and outer race oil film thicknesses were the same. For fully flooded inlet conditions the inner race film thickness is actually calculated to be 85 percent of the film thickness at the outer race. A

dielectric constant of 2 for the oil was used to prepare figure 2. During the tests, the capacitance bridge balancing signal was observed on an oscilloscope. Any short circuiting of the test bearing capacitance could be observed, which is an indication of lubricant film breakdown. A separate measurement system was used to detect percent of lubricant film with the conductance method. The concept was previously described in reference 6 and used in reference 20.

Test Procedure

Test temperatures were 27°, 65°, and 121° C (80°, 150°, and 250° F). Thrust loads applied to the bearing were 90, 445, and 778 newtons (20, 100, and 175 lb). The shaft speed was adjusted at speeds from 400 to 15 000 rpm. The general test procedure was to begin at top speed and low load for a given temperature of operation. Then the loads were increased stepwise to the heaviest load and held long enough to take a film thickness reading with the capacitance bridge. The test continued, repeating the load sequence for each successive lower speed. During the loading sequence, the shaft speed was held constant with an automatic controller. The temperature was controlled to within 3° C (5° F).

For the measurements with lubricant A the test bearing had approximately 20 to 30 hours of running time on it, which were accumulated during checkout and calibration of the test rig. For each of the remaining test lubricant sequences a new test bearing was used. The tests with the new bearings were carefully monitored to prevent continued operation in the asperity contact regime. The test sequence was stopped when it became clear that the lubricant film was breaking down. This occurred at the lower speeds and heavy loads.

RESULTS AND DISCUSSION

Film Thickness Measurements

The effect of load and speed on film thickness at three different temperatures for the four test lubricants is shown in figure 3. In all cases the oil flow was approximately the same (0.5 mL/hr) delivered by a mist; measured film thicknesses ranged from 0.025 to 0.51 micrometer (1 to 20 μin.).

Generally the measured film thickness was more weakly dependent on speed than theory suggests. At lower temperatures, where the lubricants were more viscous, film thickness was observed to be independent of speed and even decreased at the higher speeds. At the highest temperature the film thickness speed dependence was closest to theoretical prediction.

In the case of lubricants A and B, increased load gave thicker measured films (especially at 27° C), which is against the expected trend. It is not known why this was so but a speculation is offered in a later section in this report. In the case of lubricants C and D, the measured film thickness decreased with increasing load. Lubricant D gave slightly thicker films than lubricant C, as was expected, due to its higher pressure-viscosity exponent. Film thickness was the same for lubricants A and B, for the full range of temperatures, speeds, and loads. As was expected, due to the relative magnitudes of viscosity for the lubricants, the film thicknesses of A and B were significantly higher than for C and D.

Comparison with Computer Program Predicted Results

Two computer programs were used to generate predicted film thickness for the test conditions. The first program was based on the Townsend-Allen-Zaretsky EHD model (ref. 21). The computed results are shown by the solid lines in figure 3. The second program used was the NASA version SHABERTH program (refs. 22 and 23). The computed results using the SHABERTH program are shown by the dashed lines in figure 3. In all cases at the lower temperature the film thickness failed to build up with increases in speed which is contrary to the theory. In general there was only fair agreement between computer program results and experimental measurement when the data was presented and compared in figure 3 for each test condition. When the results were presented and compared in condensed form on dimensionless coordinates (figs. 4 and 5), the data at the several temperatures was unified and a definite trend of lubricant film thinning for progressively higher speeds was recognized. It was thought probable that the bearings were operating under starved conditions or that inlet shear heating was reducing the effective viscosity, in turn resulting in a thinner film. A starvation analysis was presented by Chiu (ref. 24); thermal corrections were calculated by Murch and Wilson (ref. 25) and Cheng (ref. 15). The results using these analyses are presented in another section of this text.

Comparison to Classical Theories

The data for the 90 newton (20 lb) load points were replotted on dimensionless coordinates and compared to some well-known theoretical results. The formulae chosen for comparison are attributed to Grubin (ref. 1), Dowson and Higginson (ref. 5), Archard and Cowking (ref. 4), and Hamrock and Dowson (ref. 16). These formulae are summarized as follows (Symbols are defined in the appendix.):

Grubin's formula (ref. 1):

$$H = 1.95\, G^{0.73} U^{0.73} W^{-0.091}$$

Dowson-Higginson (ref. 5):

$$H = 1.6\, G^{0.6} U^{0.7} W^{-0.13}$$

Archard-Cowking (ref. 4):

$$H = 2.04 \left(1 + \frac{2R_x}{3R_y}\right)^{-0.74} G^{0.74} U^{0.74} W^{-0.74}$$

Hamrock-Dowson (ref. 16):

$$H = 3.63(1 - e^{-0.68\,k}) G^{0.49} U^{0.68} W^{-0.73}$$

The comparisons are shown in figure 4. In general the film thickness data followed the theoretical trend with dimensionless speed for dimensionless speeds less than 10^{-10}. At the higher speeds, measured film thickness became independent of speed. This is not predicted by the theory presented.

Values of the materials parameter G in figure 4 represent the maximum, minimum, and average values for the 12 conditions of the experimental data. Since the Grubin and Archard-Cowking models both combine G and U under one exponent, the data was replotted against GU. The result is shown in figure 5. The scatter in the data was reduced by doing this.

Comparison to Other Experiments

Data from NASA sponsored experiment at the Battelle Memorial Institute are compared to the Hamrock-Dowson (ref. 16) formula and to the present experimental data in figure 6. In general, the NASA-BMI data which was taken using the X-ray film thickness measurement technique showed film thicknesses greater than theory suggests whereas the present data showed films thinner than theoretical. The synthetic paraffinic oil used in the NASA-BMI tests was the same as oils A and B and the type II ester was the same as oil C. The total amount of scatter in the data from both methods of testing is approximately equal.

In a discussion to reference 16, Kunz and Weiner present data for a steel ball sliding on a sapphire plate where film thickness was measured using the optical interferometry method. Figure 7 shows the comparison of their results to the present experimental data. Their data straddled the theoretically predicted film thickness and contained about the same scatter as the present data. The Kunz-Weiner data followed the theoretical trend with speed very well.

The present data extends the experimental points to higher speeds than previously available. The trend of dimensionless film thickness with speed does not follow the theory at the higher values of speed parameter.

Thermal Correction and Starvation Considerations

Thermal considerations in the inlet zone of the EHD zone and kinematic starvation have an effect on film reduction at higher speeds. Those theories are compared to the present data in figures 8 and 9. In figure 8 data for lubricant A and B at a 90-newton (20-lb) load was compared to theoretical results presented in reference 15. Both data and theory are for synthetic paraffinic oil. The theory assumed $G = 5000$ whereas the values for G at the three temperatures of the test as determined from table II are different for each of the six different combinations of temperature and lubricant. The average value of G was 3795 with a standard deviation of 798. Figure 8 shows that inlet shear heating may account for part of the deviation between experiment and isothermal theory at the higher speeds. However, the onset of the deviation occurs at a lower speed, indicating that another mechanism of film thickness reduction is active.

Theoretical results for starvation in a ball bearing were derived by Y. P. Chiu (ref. 24). The mechanism of lubricant distributions was assumed to be lubricant flow transverse to the rolling track. This lubricant replenishment is aided by higher lubricant-air surface tension, increased ball spacing, and reduced viscosity. Other lubrication replenishment mechanisms are also possible, depending on the particular geometric and kinematic conditions of the bearing (ref. 26). The relative effectiveness of the Y. P. Chiu replenishment model compared to other replenishment mechanisms is unknown. Starvation in roller bearings has been reported elsewhere (refs. 27 and 28).

Results from Chiu's analysis and experiments are shown in figure 9 and compared to the present data. Chiu's analysis was based on a Grubin-type analysis. Therefore, the results are presented as a function of contact lubrication flow number $G \times U$. The result of Cheng's inlet shear heating model is also shown in figure 9.

Both starvation and thermal inlet zone heating were no doubt acting during the present experiments. It is also probable that oil film replenishment was occurring

due to centrifugal or direct deposition effects. This is suggested by the continued thickness of oil film at $GU > 10^{-6}$.

Returning to figure 3 where it was noted that measured film increased with load, it is speculated that the increase in film thickness with load is due mainly to starvation effects. Perhaps the larger contact zone that occurs with the heavier load conditions is harder to starve. Therefore, the film thickness comes closest to the theoretical values when heavier loads are applied. This seems reasonable since the phenomenon seems more pronounced in the operating regime where starvation is more pronounced.

SUMMARY OF RESULTS

Film thicknesses were measured by the capacitance technique for a thrust loaded 20-millimeter-bore ball bearing. The bearing was mist lubricated at a rate of 0.5 milliliters per hour. Test parameters were thrust loads of 90, 445, and 778 newtons (20, 100, and 175 lb) which gave maximum contact stresses of 1.28, 2.09, and 2.45 gigapascals (185 000, 303 000, and 356 000 psi) on the inner race, and speeds ranging from 400 to 15 000 rpm. Tests were conducted with four different lubricants at temperatures of 27°, 65°, and 121° C (80°, 150°, and 250° F). The lubricants were (1) synthetic paraffinic, (2) synthetic paraffinic with additives, (3) a synthetic type II aircraft engine oil meeting MIL-L-23699A specifications, and (4) a synthetic cycloaliphatic hydrocarbon traction fluid. The experimental results were compared to theoretical predictions for film thickness. The following results were obtained:

1. For dimensionless speed parameter, U, greater than 10^{-10} there was gross thinning of the lubricant film. Kinematic starvation theory can account for the entire reduction in film thickness and should be used for analysis. Inlet shear heating alone does not account for the reduction in film thickness.

2. There was fair agreement between measured and theoretical film thickness at dimensionless speeds less than 10^{-10} using isothermal, fully flooded inlet EHD theory.

3. The differences between the film thicknesses calculated with the different EHD theories is less than the scatter in experimentally measured data. The significance of this is to show that some of the past refinements in film thickness prediction theory are unobservable in practice.

4. The scatter in the experimental data using the capacitance technique was the same as for data taken by other methods. This means that the general accuracy and

repeatability of measurements using X-ray, optical, and capacitance techniques are about the same.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, June 7, 1979,
505-04.

APPENDIX - SYMBOLS

a	major semi-axis of Hertz contact, m (in.)
b	minor semi-axis of Hertz contact, m (in.)
E'	reduced modulus of elasticity, $\frac{1}{E'} = \frac{1}{2}\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)$
E_1, E_2	Young's modulus for body 1 and 2, GPa (psi)
F	normal applied load, N (lb)
G	materials parameter, $\alpha E'$
H	film thickness, h/R_x
h	film thickness, m (in.)
k	ellipticity ratio, a/b
R_x	equivalent radius in rolling direction, m (in.), $\frac{1}{R_x} = \frac{1}{R_{x1}} + \frac{1}{R_{x2}}$
U	speed parameter, $u\eta_0/E'R_x$
u	average surface speed, m/sec (in./sec)
W	load parameter, $F/E'R_x^2$
x	coordinate in rolling direction, m (in.)
y	coordinate in transverse direction, m (in.)
1	body 1
2	body 2
α	pressure-viscosity exponent, GPa^{-1} (psi^{-1})
η_0	atmospheric viscosity, N-sec/m^2 (lb-sec/$in.^2$)
ν_1, ν_2	Poisson's ratio for body 1 and body 2

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TABLE I. - TEST BEARING SPECIFICATIONS

Material, balls and races	AISI M-1
Inside diameter, mm (in.)	20 (0.7874)
Outside diameter, mm (in.)	47 (1.8504)
Width, mm (in.)	14 (0.5512)
Pitch diameter, mm (in.)	35.5 (1.4)
Nominal contact angle, deg	17
Inner race curvature, percent	53
Outer race curvature, percent	54
Number of balls	3
Ball diameter, mm (in.)	7.15 (9/32)
Rockwell C hardness-inner race	62 to 64
Rockwell C hardness-outer race	62 to 64
Rockwell C hardness-balls	62 to 64
Surface finish (rms-races), μm (μin.)	0.15 (6)
Surface finish (rms-balls), μm (μin.)	0.025 to 0.05 (1 to 2)
Tolerances	ABEC-5

TABLE II. - LUBRICANTS USED IN TEST PROGRAM

Test lubricant	Manufacturer	Designation
A - Synthetic paraffinic hydrocarbon	Mobil	XRM 109F4
B - Synthetic paraffinic hydrocarbon + antiwear additive	Mobil	XRM 177F4
C - Improved type II synthetic aircraft gas turbine lubricant (MIL-L-23699A)	Texaco	SATO 7730 A2
D - Synthetic cycloaliphatic	Monsanto	Santotrac-50

TABLE III. - LUBRICANT PROPERTIES

Lubricant	A	B	C	D
Type	Synthetic paraffinic	Synthetic paraffinic	MIL-L-23699A	Traction fluid
Pour point, ^{o}C	-50	---	-60	-37
Flash point, ^{o}C	271	---	262	162
Fire point, ^{o}C	312	---		173
Autogenous ignition, ^{o}C	404	---	426	326
Density, g/cc				
100^{o} C	0.80	0.80	0.93	0.85
38^{o} C	0.84	0.84	0.98	0.89
Pressure-viscosity exponent, m^2/N				
100^{o} C	1.51×10^{-8}	1.36×10^{-8}	1.00×10^{-8}	1.70×10^{-8}
38^{o} C	1.99×10^{-8}	1.81×10^{-8}	1.37×10^{-8}	3.11×10^{-8}
Kinematic viscosity, cS				
100^{o} C (212^{o} F)	40	40	5.2	5.6
38^{o} C (100^{o} F)	447	447	28	34
Surface tension, dyne/cm				
100^{o} C	---	---	---	20
38^{o} C	31	---	---	23
Specific heat, J/(kg)(K)	2890 at 200^{o} C 2650 at 150^{o} C	--- ---	2060 at 100^{o} C 1913 at 38^{o} C	2140 at 100^{o} C 1860 at 38^{o} C
Thermal conductivity, watt/(m)(K)				
100^{o} C	---	---	0.152	0.102
38^{o} C	---	---	0.162	0.104
Additives	No	Yes[a]	Yes[b]	Yes[c]

[a]Organic phosphonate, antiwear additive.

[b]Oxidation inhibitor, corrosion inhibitor, antiwear additive.

[c]Antiwear (zinc dialkyldithiophosphate), oxidation inhibitor, antifoam, VI improver (polymethacrylate).

TABLE IV. - HERTZIAN CONTACT CONDITIONS AT INNER AND OUTER RACES FOR THREE THRUST LOADS

[Width of rolling track is determined by major axis width.]

Race	Contact condition	Thrust load, N (lbf)		
		90 (20)	445 (100)	778 (175)
Inner	Maximum Hertzian stress, GPa (ksi)	1.28 (185)	2.09 (303)	2.45 (356)
	Semimajor axis, cm (in.)	0.0510 (0.0200)	0.0840 (0.0330)	0.0990 (0.0390)
	Semiminor axis, cm (in.)	0.0066 (0.0026)	0.0110 (0.0043)	0.0130 (0.0051)
Outer	Maximum Hertzian stress, GPa (ksi)	1.13 (164)	1.85 (269)	2.19 (318)
	Semimajor axis, cm (in.)	0.0460 (0.0180)	0.0740 (0.0290)	0.0860 (0.0340)
	Semiminor axis, cm (in.)	0.0086 (0.0034)	0.0140 (0.0055)	0.0170 (0.0065)

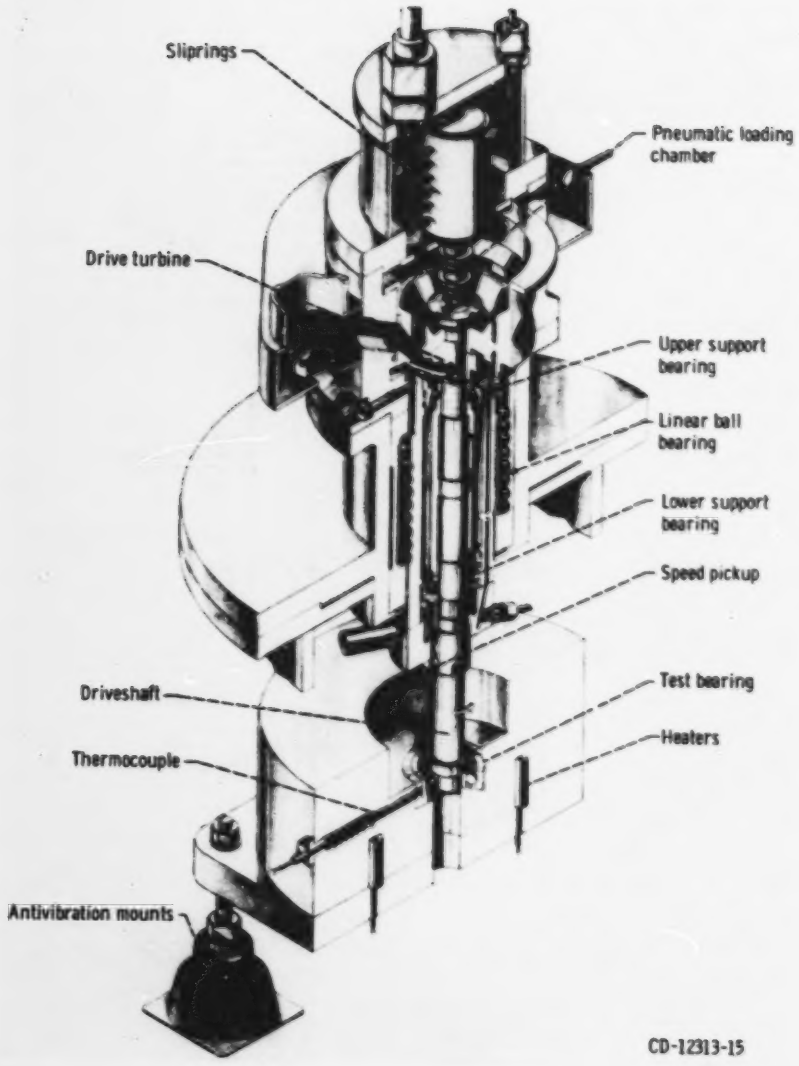


Figure 1. - EHD film thickness measurement rig.

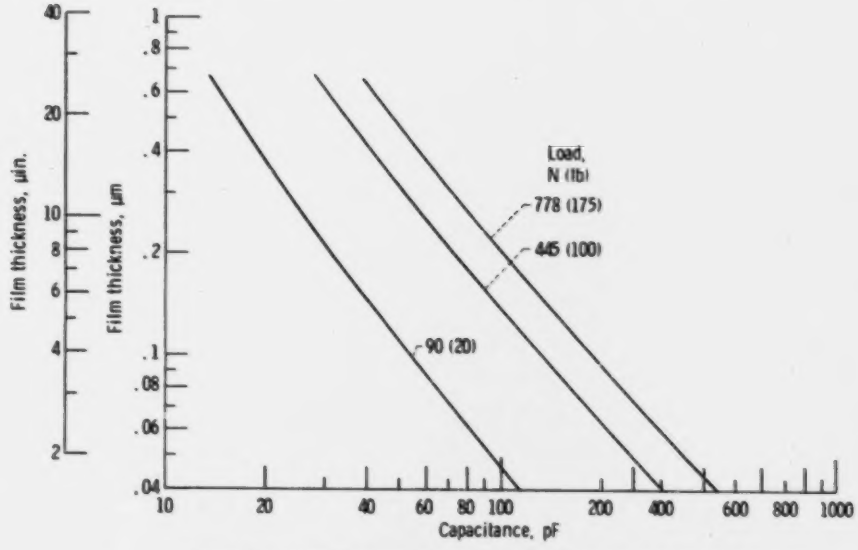


Figure 2. - Calculated capacitance for test bearing as a function of load and film thickness.

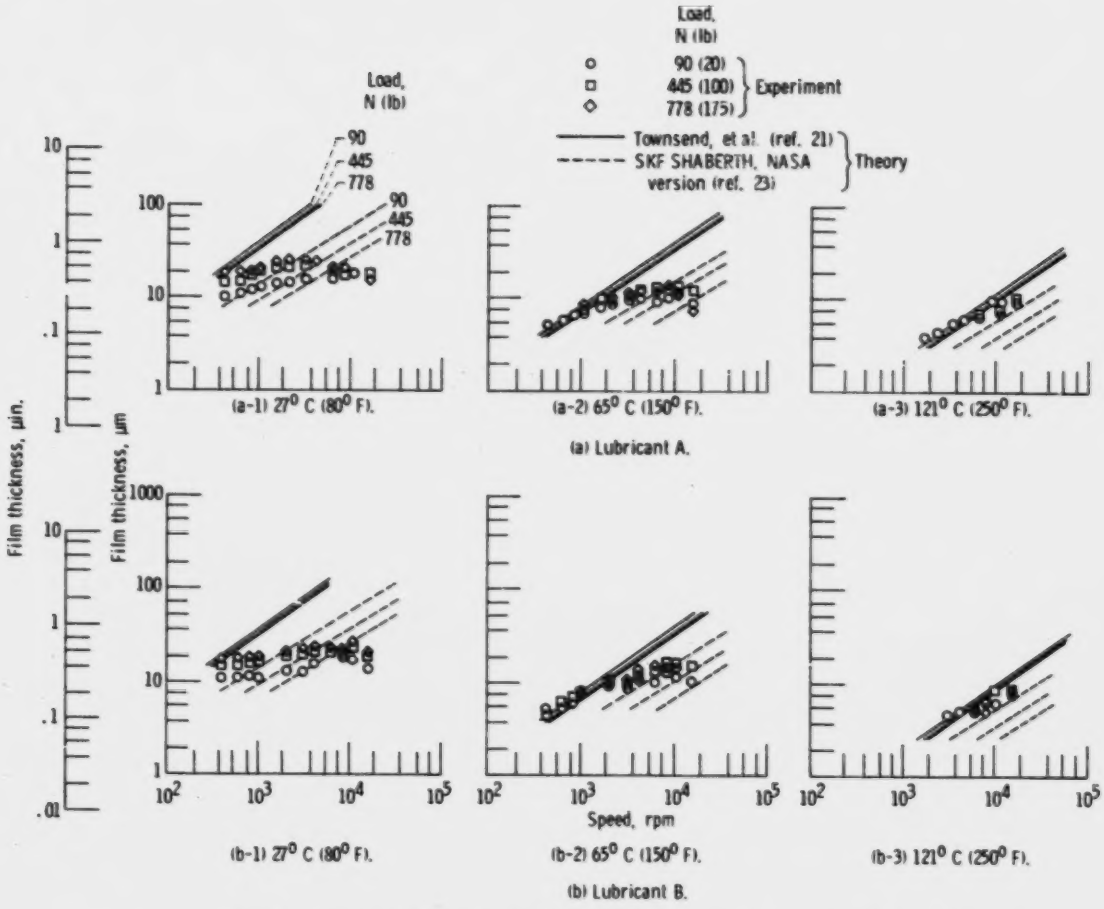
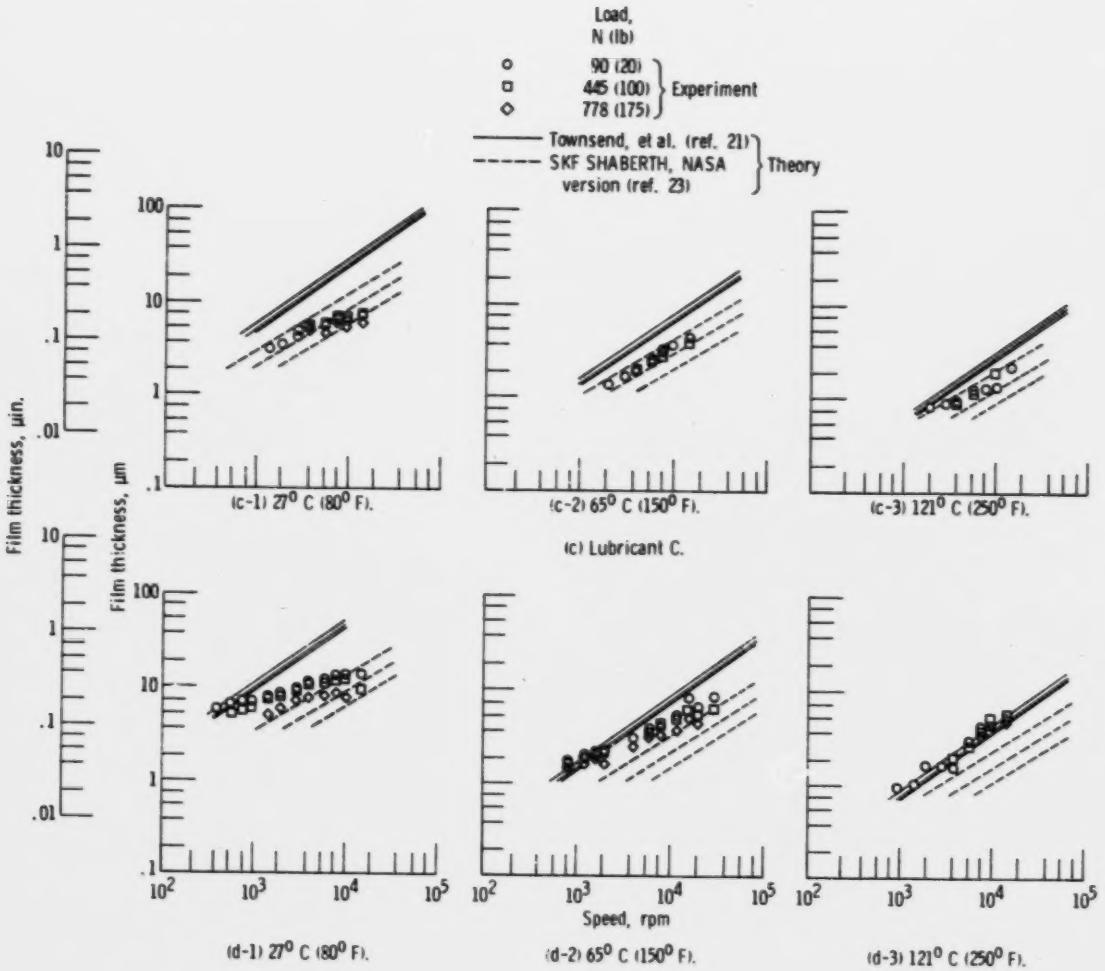


Figure 3. - Measured film thickness as a function of load, speed, and temperature for four lubricants.



(c) Lubricant C.

(d) Lubricant D.

Figure 3. - Concluded.

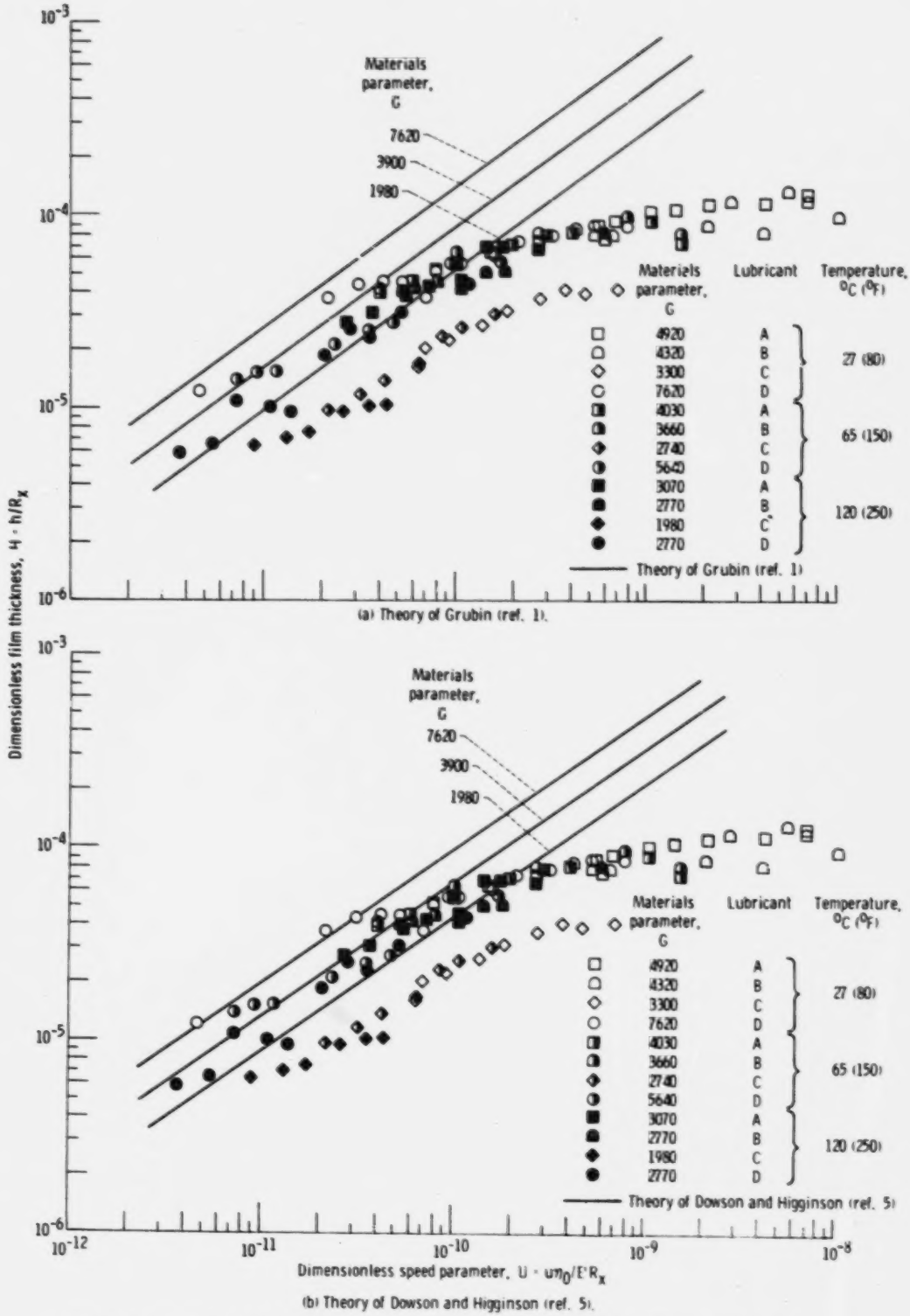
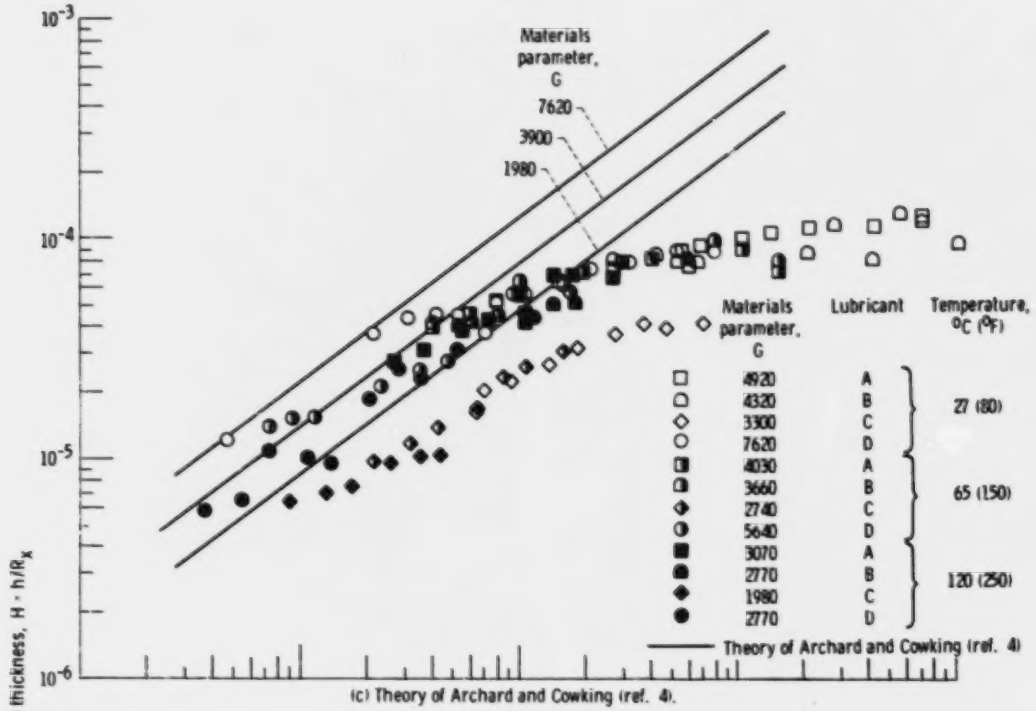
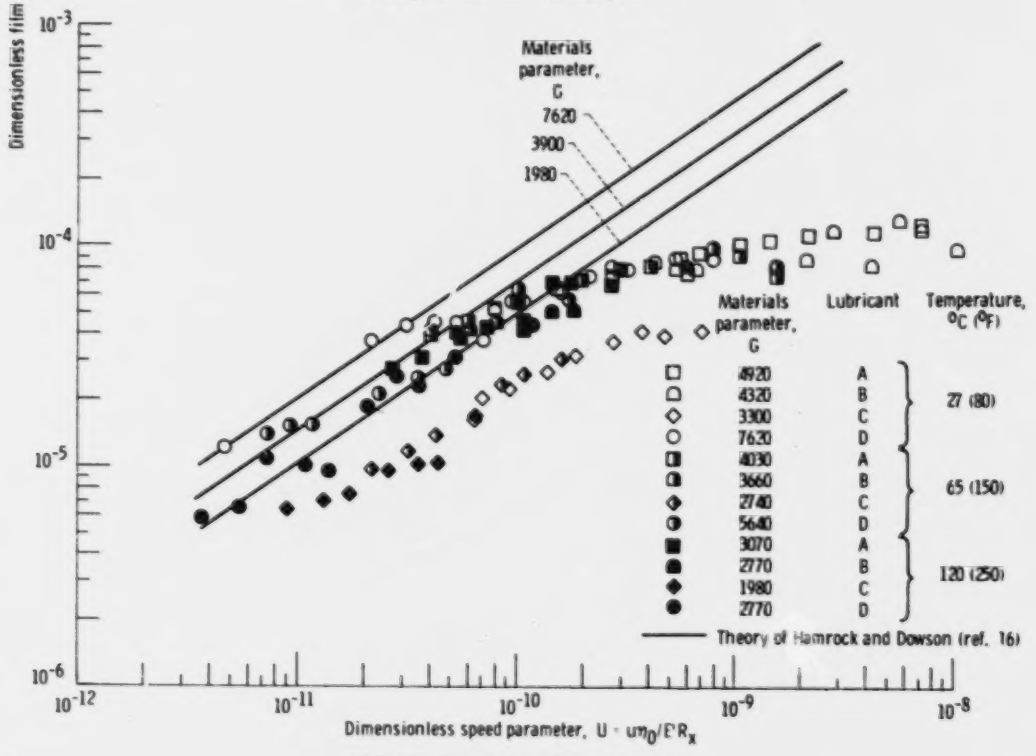


Figure 4. - Comparison of isothermal, fully flooded inlet theory with measurements for 90-newton (20-lb) load.



(c) Theory of Archard and Cowking (ref. 4).



(d) Theory of Hamrock and Dowson (ref. 16).

Figure 4. - Concluded.

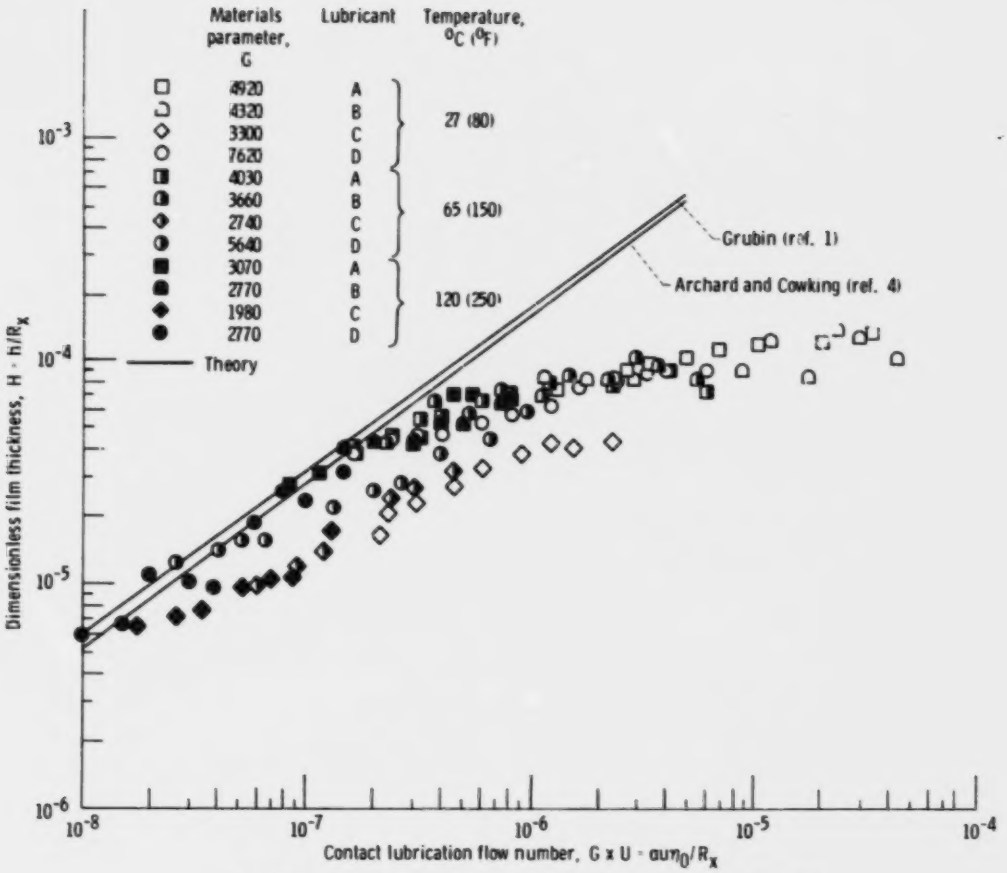


Figure 5. - Comparison of theory with experiment for 90-newton (20-lb) load.

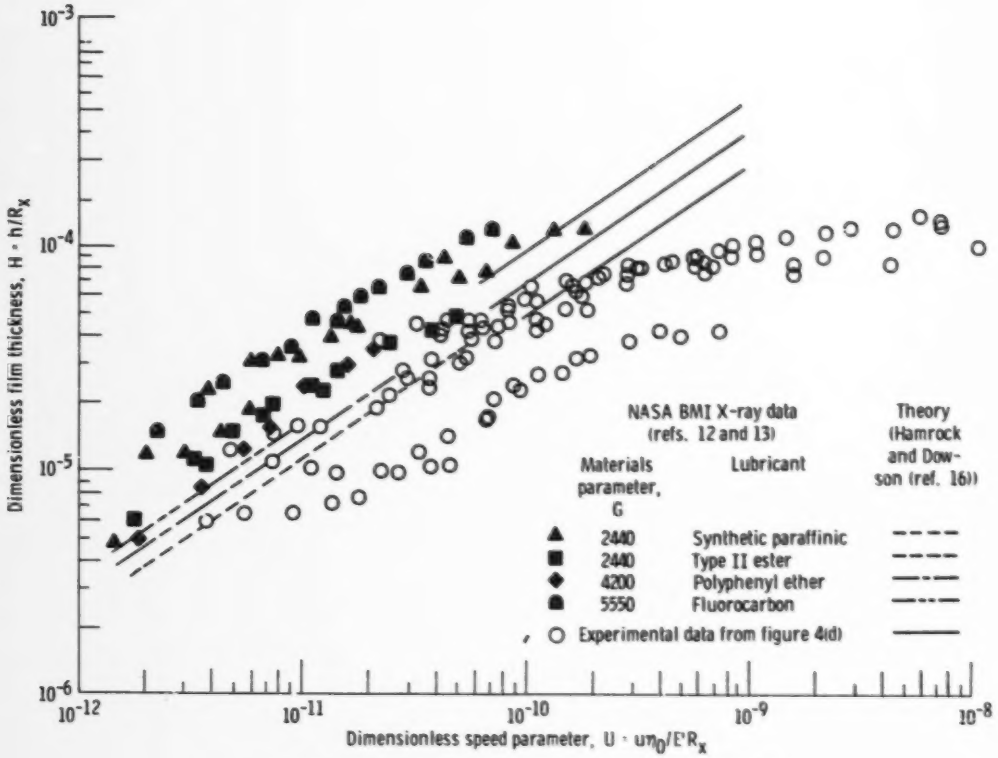


Figure 6. - Comparison with X-ray film thickness measurements.

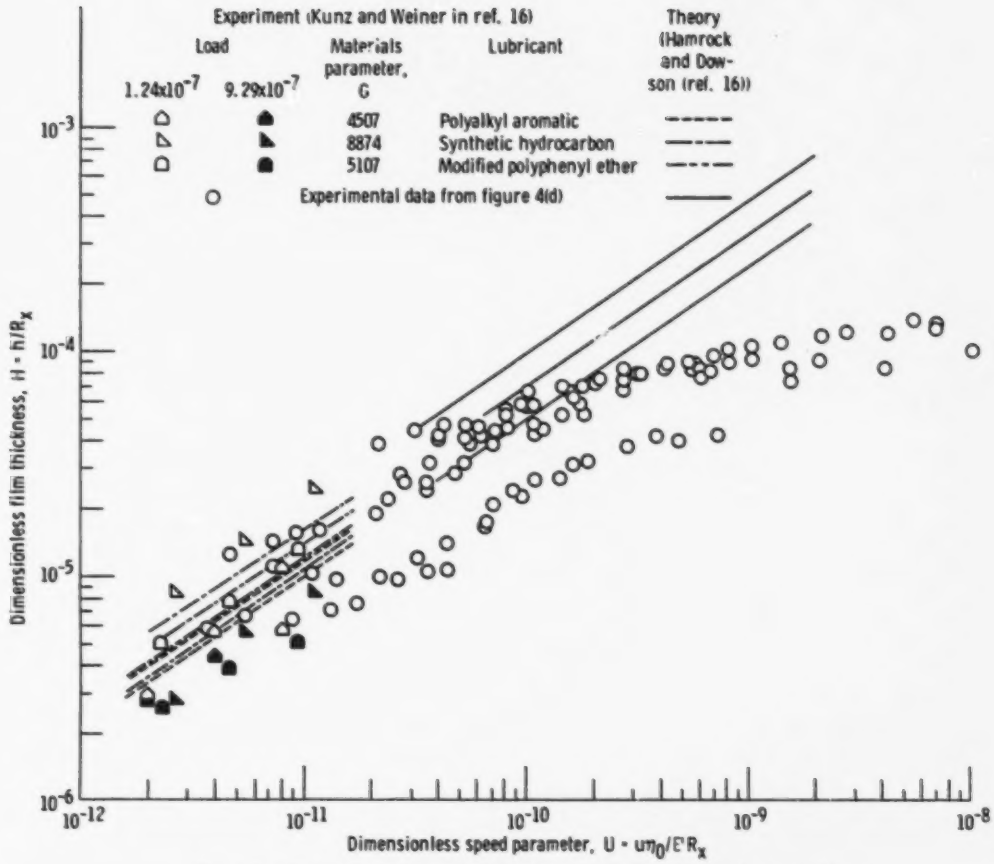


Figure 7. - Comparison with optical film thickness measurements under pure sliding.

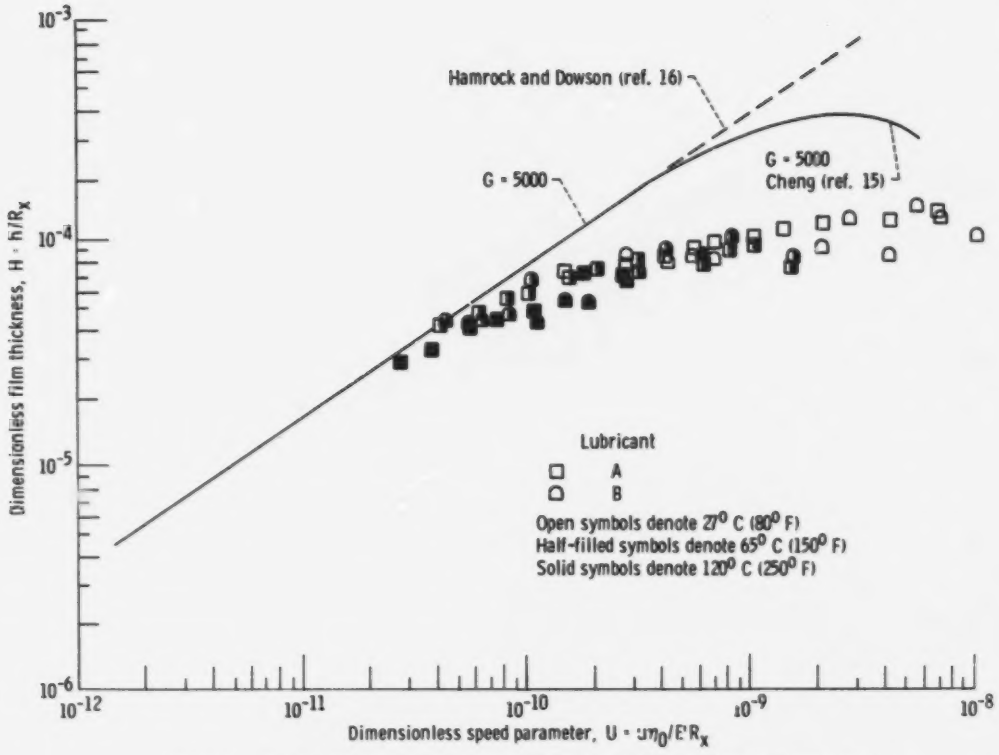


Figure 8. - Effect of inlet shear heating on film thickness and comparison with experiment.

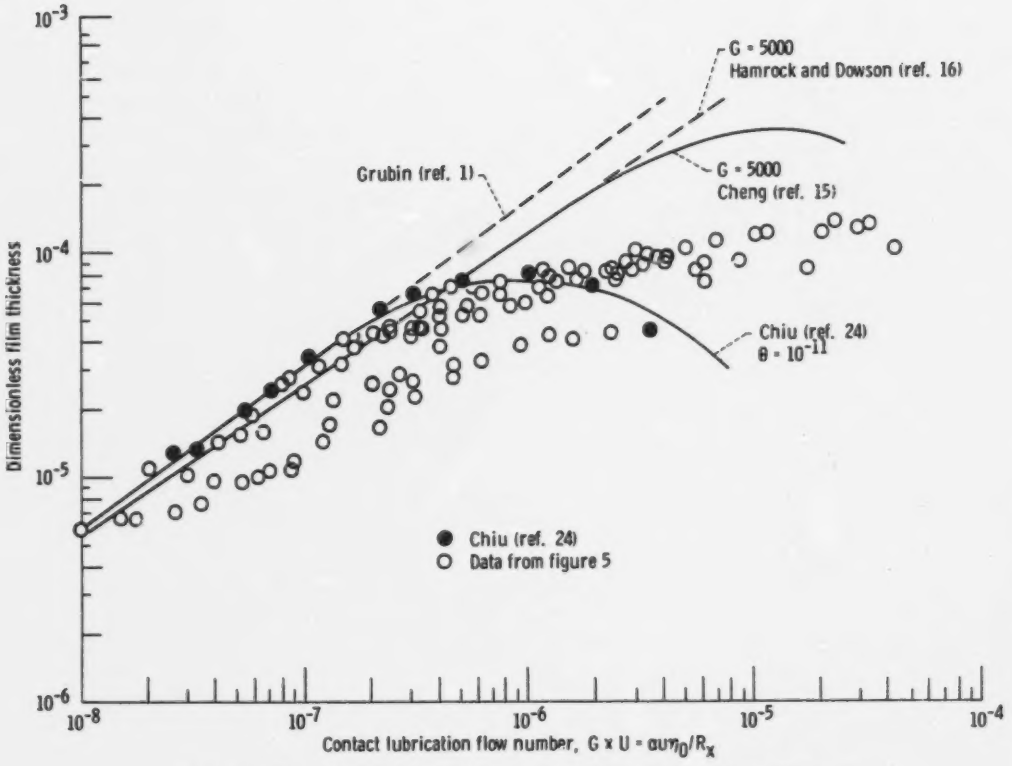


Figure 9. - Effect of kinematic starvation on film thickness and comparison with theory.

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15. Supplementary Notes

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16. Abstract

Elastohydrodynamic film thicknesses were measured for a 20-mm bore ball bearing using the capacitance technique. The bearing was thrust loaded to 90, 445, and 778 N (20, 100, and 175 lb). The corresponding maximum contact stress on the inner race was 1.28, 2.09, and 2.45 GPa (185 000, 303 000, and 356 000 psi). Test speeds ranged from 400 to 15 000 rpm. Measurements were taken with four different lubricants: (1) synthetic paraffinic, (2) synthetic paraffinic with additives, (3) synthetic type II aircraft engine oil meeting MIL-L-23699A specifications, and (4) synthetic cycloaliphatic hydrocarbon traction fluid. The test bearing was mist lubricated. Test temperatures were 27°, 65°, and 121° C (80°, 150°, and 250° F). The measured results for the various test parameters were compared to theoretical predictions from computer programs. Also the data were plotted on dimensionless coordinates and compared to several classical isothermal theories. Agreement with the computer program results was only fair. Agreement was best at low speeds and higher temperature. Plotting the results on dimensionless coordinates unified the data. There was good agreement at low dimensionless speed, but the film was much thinner than theory predicted at higher speeds. It is shown that inlet shear heating partially accounts for the film thickness reduction, whereas starvation is shown to be the primary factor.

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